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METHOD OF CALCULATING TRANSIENTS IN GAS TURBINE POWER PLANTS

V. V. Selin

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METHOD OF CALCULATING TRANSIENTS IN GAS TURBINE POWER PLANTS

by

V. V. Selin

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By: V. V. Selin

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ABSTRACT: A method of finite differences for calculating the transient characteristics of a gas-turbine power plant is offered; the method is claimed to be "simpler and more reliable as compared to existing methods." The following formulas are developed: rate-of-flow in the turbine; same in the compressor; compression in the compressor; expansion in the turbine; balance of total compression ratio and total expansion ratio; rate-of-change of gas quantity in the capacity (regeneration case); heat exchange in the regenerator; power balance in the compressor shaft; compressor efficiency; turbine efficiency. The above formulas, some additional turbine formulas, and ship-propulsion formulas are tabulated. Peculiarities in the calculation of a load-drop case are discussed. Orig. art. has: 2 figures, 33 formulas, and 1 table. English Translation: 12 pages.

U. S. BOARD ON GEOGRAPHIC NAMES TRANSLITERATION SYSTEM

Block	Italic	Transliteration	Block	Italic	Transliteration
А а	*А а*	A, a	Р р	*Р р*	R, r
Б б	*Б б*	B, b	С с	*С с*	S, s
В в	*В в*	V, v	Т т	*Т т*	T, t
Г г	*Г г*	G, g	У у	*У у*	U, u
Д д	*Д д*	D, d	Ф ф	*Ф ф*	F, f
Е е	*Е е*	Ye, ye; E, e*	Х х	*Х х*	Kh, kh
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З з	*З з*	Z, z	Ч ч	*Ч ч*	Ch, ch
И и	*И и*	I, i	Ш ш	*Ш ш*	Sh, sh
Й й	*Й й*	Y, y	Щ щ	*Щ щ*	Shch, shch
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Н н	*Н н*	N, n	Э э	*Э э*	E, e
О о	*О о*	O, o	Ю ю	*Ю ю*	Yu, yu
П п	*П п*	P, p	Я я	*Я я*	Ya, ya

* ye initially, after vowels, and after ъ, ь; e elsewhere. When written as ё in Russian, transliterate as yё or ё. The use of diacritical marks is preferred, but such marks may be omitted when expediency dictates.

METHOD OF CALCULATING TRANSIENTS IN GAS TURBINE POWER PLANTS

Engineer V. V. Selin

Kaliningrad Mechanical Institute of Fishing Industry and Economy

For a well-founded selection of a gas turbine power plant [GTU](ГТУ) layout, during its designing of great value is a detailed calculated analysis of characteristics in transitional modes (transients), for which a sufficiently simple and exact method of calculation is necessary. In this article a method of calculation is offered which is more simple and reliable, as compared to the existing methods.

Under variable conditions, the number of independent parameters for any layout of a GTU is equal to the number of combustion chambers [1]. During transitions of a GTU, the number of independent parameters, simply determining this or that mode, is equal to the sum of the number of working combustion chambers and the number of rotors in the layout. As an example let us examine the two-shaft layout with a low pressure power turbine [TND](ТНД) (Fig. 1). If for this layout we assign, for example, the temperature beyond the combustion chamber and the number of rotor turns, then we will simply determine the mode of the power plant.

Data available in literature permit examining the layout of a high pressure air duct as one which is discrete [3, 5], i.e., to consider that the given capacity is concentrated in one point — beyond the regenerator (Fig. 2).

One may assume that establishing the gas parameters before the turbine and after it occurs instantly [2, 3], which is equivalent to disregarding the gas volumes in the flow-through part of the turbine. Here the working substance can be considered as being inertialess. This assumption will agree with the results of experimental investigations [3].

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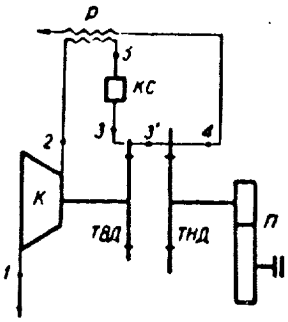

Fig. 1. Diagram of a two-shaft ГТУ with independent low pressure power turbine. ТВД — high pressure turbine; ТНД — low pressure turbine; К — compressor; Р — regenerator; КС — combustion chamber; П — reductor; 1, 2... — indices of the parameters of working substance at a given point.

When determining time τ during the transition from one established mode to another the parameters of these conditions and character of load on power shaft are assumed to be well-known.

The problem is solved with the help of the method of finite differences. One may assume that the entire transient process consists of a number of unsteady modes of the power plant, each of which is included in a certain small, finite interval of time $\Delta\tau_1$.

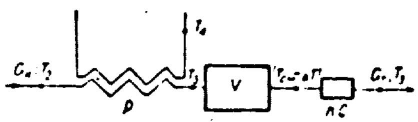

Fig. 2. Diagram of high pressure gas-air duct: Р — regenerator; КС — combustion chamber; V — given capacity (tank).

As the condition which disturbs the balanced state we assume the assigned law of change of gas temperature beyond the combustion chamber $T_3 = f(\overline{G}_T)$, taking into account the dynamic characteristics of the control system. Here and below by a dash we have designated the relative magnitudes, i.e., $\overline{G}_T = G_T/G_{T0}$ — the flow rate ratio through the turbine in a given mode to the flow rate in the nominal mode. Numerical index at parameters T and p corresponds to the designations in Fig. 1.

For determining the parameters of each unsteady mode, it is necessary to solve a system of equations which characterizes the work of all basic ГТУ elements.

<u>Equation of the flow rate for a turbine</u> in general has the form

$$G_t = f(p_{30};\ p_{40};\ T_{30};\ n_0;\ p_3;\ p_4;\ T_3;\ n),$$

where p_3 and p_4 are the pressure before and beyond the turbine; T_3 is the temperature of gas before the turbine; n is the number of revolutions.

For multistage turbines it is possible to use the Stodol equation, and for low stage turbines when determining flow rate, we can use the method of I. V. Kotlyar [1]. Using the flow rate equation, for the assigned law of change of gas temperature before the turbine $T_3 = f(\overline{G}_T)$, it is possible to construct a characteristic of the turbine $p_3 = f(\overline{G}_T)$.

Equation of flow rate for a compressor is assumed to be assigned in the form of a universal characteristic.

Equation of compression in the compressor

$$L_к = \frac{H_к}{\eta_к} = \frac{c_p T_1 (\varepsilon^{\frac{k-1}{k}} - 1)}{\eta_к}, \quad (1)$$

where $L_К$ is the specific work of compression in the compressor; $H_К$ is the isoentropic heat drop in the compressor; T_1 is the air temperature at the entrance to the compressor; ε is the compression ratio; $\eta_К$ is the internal efficiency of the compressor; k is the index of isoentropy.

Air temperature beyond the compressor is

$$T_2 = T_1 + \frac{H_к}{c_p \eta_к}. \quad (2)$$

Equation of expansion for a turbine is

$$L_т = H_т \eta_т = \eta_т c_p T_3 \left(1 - \frac{1}{\varphi^{\frac{k-1}{k}}}\right). \quad (3)$$

where L_T is the specific work of expansion in the turbine; H_T is the isoentropic heat drop in the turbine; η_T is the internal efficiency of the turbine; c_p is the specific heat capacity of gas; φ is the expansion ratio.

Gas temperature beyond the turbine

$$T_4 = T_3 - \frac{H_т \eta_т}{c_p}; \quad (4)$$

$H_К$ and H_T are determined by the thermal diagram [1].

In certain cases one should consider the nonstationary heat exchange in the flow-through part of the turbine.

The heat which is imparted by the gas to the metal (by the metal to the gas) during the time $\Delta\tau_i$,

$$\Delta Q = \frac{\alpha F \Delta\tau_i}{3600} (T_i - T_{ст\, i}). \quad (5)$$

where α is the average (along the flow-through part) coefficient of heat transfer; F is the surface, washed by the gas; $T_Г$ and T_{CT} are, correspondingly, the average temperatures of gas and the walls of the blade channels.

A change of gas temperature

$$\Delta T_{г} = \frac{\Delta Q}{G_{т} c_{p} \Delta\tau_{i}}. \tag{6}$$

A change of temperature of the channel walls

$$\Delta T_{ст} = \frac{\Delta Q}{G_{м} c_{м}}, \tag{7}$$

where G_M and c_M are, correspondingly, the weight of metal, participating in the heat exchange, and its specific heat capacity.

As a result the temperature of gas beyond the turbine will be determined

$$T_4' = T_4 - \Delta T_{г}. \tag{8}$$

A change of heat drop in the turbine as a result of the heat exchange of gas with metal will be proportional to the change of temperature

$$\Delta H = \frac{H_{т} \Delta T_{г}}{2T_3}. \tag{9}$$

The influence of thermal accumulation in the metal of the combustion chamber and the gas conduits can be disregarded due to the low coefficients of heat transfer and the low metal content of the combustion chamber. In many cases it 's also possible not to consider the heat exchange in the flow-through part.

<u>Equation of balance of the general compression ratio and expansion is</u>

$$\varphi_{общ} = \xi \pi_{общ}, \tag{10}$$

where ξ is the loss factor of pressure.

<u>Equation of the speed of change of the quantity of gas in capacity</u>. In case of the presence of regeneration and the gas-air capacities connected with it, flow rate through the turbine in unsteady modes will be different than the flow rate through the compressor. This difference in flow rates $\Delta\overline{G}_i$; at the end of any assigned interval of time $\Delta\tau_i$ will be directly proportional to the increase of pressure p and inversely proportional to the increase of gas temperature T in the tank. As a result,

$$\Delta\overline{G}_i = \frac{R_{v(i-1)} \overline{G}_{т(i-1)}}{\Delta\tau_i} \left(\frac{\pi_i T_{(i-1)}}{\pi_{(i-1)} T_i} - 1 \right), \tag{11}$$

where index i pertains to the desired mode, while (i - 1) pertains to the preceding mode; R_V is time of capacity, i.e., the time in which the tank V will be emptied in the presence of flow rate $\overline{G}_T$.

With a sufficient degree of accuracy, the temperature of gas in the tank can be determined as the temperature of air beyond the regenerator $T_1 = T_5$. Value ε_1 can be found by the trial-and-error method, by the characteristics of the compressor and the turbine.

<u>Equation of heat exchange in the regenerator.</u> As has been already noted by many authors [2, 4], the accumulation of heat in the metal of a regenerator renders a very significant influence on transitions of a ГТУ. This should be considered when determining the air temperature beyond regenerator T_5.

The heat which is imparted by the gas to the regenerator wall (by the wall to the gas),

$$\Delta Q_{\Gamma} = \frac{\alpha_{\Gamma} F \Delta\tau_i}{3\,600} (T_{\Gamma} - T_{\text{ст}}). \tag{12}$$

the heat, imparted by the regenerator wall to the air (air to wall),

$$\Delta Q_{\text{в}} = \frac{\alpha_{\text{в}} F \Delta\tau_i}{3\,600} (T_{\text{ст}} - T_{\text{в}}). \tag{13}$$

A change of thermal potential of the wall

$$\Delta Q_{\text{ст}} = \Delta Q_{\text{в}} - \Delta Q_{\Gamma}. \tag{14}$$

In equations (12) and (13), α_{Γ} and α_{B} are coefficients of heat transfer from the gas to the wall and from the wall to the air, which in the transient process can be, with a sufficient degree of accuracy, taken as depending only on the flow rates of gas and air, if we were to consider the low variability of coefficients of thermal conduction λ and coefficients of viscosity μ:

$$\begin{aligned} \alpha_{\Gamma} &= \alpha_{\Gamma 0}\, \bar{G}_{\text{т}}^{m}; \\ \alpha_{\text{в}} &= \alpha_{\text{в}0}\, \bar{G}_{\text{к}}^{n}. \end{aligned} \tag{15}$$

where m and n are indices of degrees which enter into equation $Nu = a\ Re^{m}$ and depend on the type of heat exchanger; F is the surface of the heat exchanger; T_{CT}, T_{Γ}, T_{B} are averaged temperatures of the wall, gas and air.

Gas temperature beyond the regenerator is

$$T_6 = T_4 - \frac{\Delta Q_{\Gamma}}{G_{\text{т}} c_p \Delta\tau_i}. \tag{16}$$

Air temperature beyond the regenerator is

$$T_5 = T_2 + \frac{\Delta Q_{\text{в}}}{G_{\text{к}} c_p \Delta\tau_i}. \tag{17}$$

A change of wall temperature of the regenerator

$$\Delta T_{\text{ст}} = \frac{\Delta Q_{\text{ст}}}{G_{\text{м}} c_{\text{м}}}. \tag{18}$$

where G_M and c_M are, correspondingly, the weight of metal, participating in the heat exchange, and its specific heat capacity.

Value ΔT_{CT} is usually low, and in many cases the temperature of the wall of the heat exchanger in transient processes can be considered as being constant.

Equation of balance of powers of the compressor shaft. Excess moment of the compressor shaft

$$\Delta M_{к} = M_{ТВД} - M_{к} - M_{тр}, \tag{19}$$

where $M_{ТВД}$, $M_{К}$, $M_{ТР}$ are moments of the turbine, compressor and friction.

A change of revolutions of the compressor shaft

$$\Delta n_{к} = \frac{30}{\pi I_{к}} \Delta M_{к} \Delta \tau_{i}, \tag{20}$$

where $I_{К}$ is the moment of inertia of the compressor shaft.

Equation of balance of powers of the power shaft. Excess moment of the power shaft

$$\Delta M_{c} = M_{ТНД} - M_{дв} - M_{тр}. \tag{21}$$

A change of revolutions of the power shaft

$$\Delta n_{c} = \frac{30}{\pi I_{c}} \Delta M_{c} \Delta \tau_{i}, \tag{22}$$

where I_C is the moment of inertia of the power shaft (including the reductor and the consumer), applied to revolutions of the power turbine. $M_{ДВ}$ is the moment on the consumer shaft, which can be determined by characteristics, taking into account the delay of the control system. For example, for a variable pitch screw (ВРШ) it is sufficient to have dependences

$$\left.\begin{aligned} M_{дв} &= f\left(\bar{n}_{c};\ \frac{H}{D};\ v\right); \\ P_{в} &= f\left(\frac{H}{D};\ v\right); \\ R &= f(v), \end{aligned}\right\}$$

applied to revolutions of the power turbine, and a program of screw pitch variation.

Here H/D is the pitch ratio; v is the speed of vessel in knots; R is water drag on vessel motion; P_B is total thrust of the screws.

A change of speed of a vessel for the given interval of time $\Delta\tau_i$ will be determined from equation

$$m\frac{dv}{dt}=P_{e}-R, \tag{23}$$

where m is vessel mass with additional mass of the water. Internal efficiency of the compressor can be found by its characteristic.

Efficiency of the turbine can be calculated by well-known dependences $\eta_{0Л} = f(x)$ in the variable mode, if $x = \frac{u}{c_0}$ is defined as the given ratio of speeds

$$x=x_0\bar{n}\sqrt{\frac{H_{T0}}{H_T}}.$$

After determining all the parameters of the i-th mode, we should set as our goal the time interval $\Delta\tau_{(i+1)}$ and find parameters of the (i + 1)-th mode. The calculation should be conducted up to achievement of the parameters of assigned conditions. An example of the calculation sequence is represented in Table 1, where two unsteady modes are determined during reception of a load for the ГТУ (Fig. 1), working on ВРШ. The law of temperature change before the turbine is accepted as

$$T_3=\text{const}=1\,098^{\circ}\,\text{K},$$

which indirectly considers the accumulation of heat in the regenerator.

According to calculated data it is possible to construct dependence $\bar{n} = f(\tau)$ and the line of modes of the compressor in the transient process. Time of the transient process τ will be determined by the sum of time intervals accepted in this calculation

$$\tau=\sum_{1}^{n}\Delta\tau_i. \tag{24}$$

The given method can be applied not only to the simplest, but also to any complicated layouts of ГТУ. In the presence of layouts of ГТУ of several compression ratios, division of compression ratios by steps should be made by proceeding from the condition of their joint work [1]

$$B_j=B_{j0}\frac{\varepsilon_{(j-1)0}}{\varepsilon_{(j-1)}}\sqrt{\frac{T_j}{T_{j0}}}, \tag{25}$$

where B is the parameter of flow rate, equal to $\frac{Gp_0}{G_0p}\sqrt{\frac{T}{T_0}}$; j is the ordinal number of the compression ratio.
Here

$$\varepsilon=\varepsilon_1\varepsilon_2\varepsilon_3\ldots. \tag{26}$$

Table 1

Designation of value	Formula	Dimension	Value	
			i	$i+1$
Interval of time	τ -- we assign	s	10	10
Number of revolutions of compressor shaft	$n_{\text{к}} = n_{\text{к}(i-1)} + \Delta n_{\text{к}(i-1)}$	r/min	2666	2912
Relative number of revolutions of compressor shaft	$\bar{n}_{\text{к}} = \frac{n_{\text{к}}}{n_{\text{к}0}}$	—	0,444	0,49
Compression ratio	ε — we assign	—	1,65	1,8
	$\Delta\bar{G} = \frac{R_{t(i-1)}\bar{G}_{t(i-1)}}{\Delta\tau} - \left(\frac{\varepsilon T_{5(i-1)}}{\varepsilon_{(i-1)}T_3}\right)$	--	0,0075	0,008
Relative flow rate through the compressor	$\bar{G}_{\text{к}} = f(\bar{n}_{\text{к}}; \varepsilon)$ (by characteristic	--	0,2575	0,313
Relative flow rate through the turbine	$\bar{G}_{\text{т}} = \bar{G}_{\text{к}} - \Delta\bar{G}$	—	0,25	0,305
Pressure beyond the КНД	$p_2 = p_3 + f(\bar{G}_{\text{т}})$	kg/cm^2	1,65	1,8
Compression ratio (check)	$\varepsilon = \frac{p_2}{p_1}$	—	1,65	1,8
Pressure beyond the ТВД	$p_3' = \sqrt{p_3^2 - \bar{T}_3\bar{G}_{\text{т}}^2(p_{30}^2 - p_{30}'^2)}$	kg/cm^2	1,094	1,14
Expansion ratio of ТВД	$\varphi_1 = \frac{p_3}{p_3'}$	—	1,51	1,58
Heat drop of ТВД	$H_1 = f(\varphi_1; t_3)$	kcal/kg	29,5	33,2
Coefficient	$x_1 = x_0\bar{n}_{\text{к}}\sqrt{\frac{H_{10}}{H_1}}$	—	0,312	0,32
Efficiency of ТВД	$\eta_1 = f(x_1)$	—	0,685	0,7
Power of ТВД	$N_1 = 4{,}19\, G_0\bar{G}_{\text{т}}H_1\eta_1$	kW	1176	1525
Moment of ТВД	$M_1 = 973\frac{N_1}{n_{\text{к}}}$	kg-m	430	505
Efficiency of compressor	$\eta_{\text{к}}$ — by characteristic	—	0,78	0,76
Heat drop of compressor	$H_{\text{к}} = f(\varepsilon; t_1)$	kcal/kg	10,6	12,7
Air temperature beyond the compressor	$T_2 = T_1 + \frac{H_{\text{к}}}{c_p\eta_{\text{к}}}$	°K	344	355
Power of the compressor	$N_{\text{к}} = 4{,}19\, G_0\bar{G}_{\text{к}}\frac{H_{\text{к}}}{\eta_{\text{к}}}$	kW	834	1124
Moment of the compressor	$M_{\text{к}} = 973\frac{N_{\text{к}}}{n_{\text{к}}}$	kg-m	305	374
Excess moment of the compressor shaft	$\Delta M_{\text{к}} = M_1 - M_{\text{к}} - M_{\text{тр}}$	kg-m	75	81

Table 1 (continued)

Designation of value	Formula	Dimension	Value	
			i	$i+1$
Change of revolutions of compressor shaft	$\Delta n_{к} = \frac{30}{\pi I_{к}} \Delta M_{к} \Delta\tau$	r/min	276	298
Number of revolutions of power shaft	$n_c = n_{c(i-1)} + \Delta n_{c(i-1)}$	r/min	1700	1951
Relative number of revolutions of power shaft	$\bar{n}_c = \frac{n_c}{n_{c0}}$	—	0,34	0,39
Expansion ratio of ТНД	$\varphi_2 = \frac{p_3'}{p_4}$	—	1,094	1,14
Temperature before ТНД	$T_3' = T_3 - \frac{H_1\eta_1}{c_p}$	°K	1023	1012
Heat drop of ТНД	$H_2 = f(t_3';\ \varphi_2)$	kcal/kg	6	9
Coefficient	$x_2 = x_0 \bar{n}_c \sqrt{\frac{H_{20}}{H_2}}$	—	0,451	0,432
Efficiency of ТНД	$\eta_2 = f(x_2)$	—	0,83	0,82
Power of ТНД	$N_2 = 4{,}19\, G_0 \bar{G}_T H_2 \eta_2$	kW	268	477
Moment of ТНД	$M_2 = 973 \frac{N_2}{n_c}$	kg-m	149	238
Speed of the vessel	$v = v_{(i-1)} + \Delta v_{(i-1)}$	knots	0,05	0,08
Relative screw pitch	$\frac{H}{D}$ — by program	—	0,3	0,3
Moment of the screw	$M_{в} = f\left(\bar{n}_c;\ \frac{H}{D};\ v\right)$	kg-m	40	52
Excess moment of power shaft	$\Delta M_c = M_2 - M_{в} - M_{тр}$	»	64	141
Change of revolutions of power shaft	$\Delta n_c = \frac{30}{\pi I_c} \Delta M_c \Delta\tau$	r/min	251	561
Thrust of the screw	$P_{в} = f\left(\frac{H}{D};\ v\right)$	kg	3000	5500
Water drag to vessel travel	$R = f(v)$	»	0	0
Change of vessel speed	$\Delta v = \frac{P_{в} - R}{0{,}515\, m} \Delta\tau$	knots	0,03	0,055
Temperature beyond the ТНД	$T_4 = T_3' - \frac{H_2\eta_2}{c_p}$	°K	1005	989

Table 1 (continued)

Designation of value	Formula	Dimension	Value	
			i	$i+1$
Temperature beyond the regenerator[1]	$T_5 = T_2 + r(T_4 - T_2)$	°K	807	799
Specific gravity of gas in the tank	$\gamma = 341{,}5\frac{p_3}{T_5}$	kg/m³	0,698	0,77
Time of capacity (tank)	$R_v = \frac{R_{t0}\gamma}{G_T\gamma_0}$	•	3,07	3

[1]In this case temperature T_5 is determined by a simplified formula, since it is used only when determining γ and R_v; r is the degree of regeneration.

In case of a ГТУ layout without regeneration, the calculation is simplified. For layouts with intermediate heat feed, it is necessary to assign a law of change of gas temperature in the intermediate combustion chamber.

<u>Peculiarities of calculating ГТУ layouts for load-drop.</u> In order to determine the dynamic stoking of ГТУ revolutions during full load-drop (for example, cutoff of a generator from the network), it is necessary to know the value of time lag of the control system τ_p (the time which has passed from load-drop up to cessation of fuel feed). After cessation of fuel feed, pressure of gas before the turbine will be determined by the pressure of gas in the tank. For the case of isoentropic expansion, the connection between the parameters of gas in the tank will be expressed as

$$p\gamma^{-k} = p_0\gamma_0^{-k},$$

where γ is the specific gravity of gas.

By differentiating this equation we will obtain

$$dp = \frac{p_0 k \gamma^{k-1}}{\gamma_0^k} d\gamma.$$

In the presence of small changes of p, ratio

$$\frac{\gamma^{k-1}}{\gamma_0^{k-1}} \approx 1.$$

Thus,

$$d\gamma = dp\frac{\gamma_0}{p_0 k}.$$

A change of weight quantity of gas in the tank can be determined as

$$d\Pi = Vd\gamma,$$

or

$$d\Pi = (G_т - G_к)\,d\tau.$$

If we consider that in the normal mode $G_{TO} = G_{KO}$, then, crossing to finite increments, after certain transformations we can determine the relative change of pressure in the tank as a result of expansion for the time interval $\Delta\tau_1$.

$$\Delta\bar{p} = \frac{k\Delta\tau_i}{R_p}(\bar{G}_т - \bar{G}_к). \tag{27}$$

One can assume that after cessation of fuel feed into the combustion chamber, gas temperature before turbine T_3 at any moment of time is equal to the exit temperature from the tank. In turn, gas temperature in the tank at the end of time interval $\Delta\tau_1$ will be determined by the temperature of displacement T_{CM} of gas which is anew proceeding to the tank from the regenerator, during the time $\Delta\tau_1$ with the remaining gas in the tank and the change of gas temperature as a result of expansion ΔT

$$T_3 = T_{см} - \Delta T. \tag{28}$$

If during interval of time $\Delta\tau_1$ into the tank quantity of gas $\bar{G}_K\Delta\tau_1$ with temperature T_5 proceeded from the regenerator, and from the tank quantity of gas $\bar{G}_T\Delta\tau_1$ with temperature T_3 departed into the turbine, then from the equation of heat balance, if one were to accept the process of mixing as being instantaneous, we can obtain the temperature of mixing in the tank

$$T_{см} = \frac{(R_p - \Delta\tau_i)\bar{G}_т T_3 + \Delta\tau_i \bar{G}_к T_5}{(R_p - \Delta\tau_i)\bar{G}_т + \Delta\tau_i \bar{G}_к}. \tag{29}$$

A change of gas temperature in the tank, as a result of isoentropic expansion, will be expressed by equation

$$\bar{T} = (\bar{p})^{\frac{k-1}{k}},$$

or

$$d\bar{T} = \frac{k-1}{k}\bar{p}^{-\frac{1}{k}}dp.$$

Considering that in the presence of small changes of pressure value $\bar{p}^{-\frac{1}{k}} \approx 1$, by crossing to finite increments, we can obtain

$$\Delta\bar{T} = \frac{k-1}{k}\Delta\bar{p}. \tag{30}$$

For any intermediate mode the capacity time is

$$R_{\nu} = \frac{R_{\nu 0} \gamma}{\bar{G}_{\text{т}} \gamma_0}. \tag{31}$$

According to the data of detailed calculation for load-drop, it is possible to construct dependence $\bar{n} = f(\tau)$ and to determine the value of dynamic stoking of turns $\Delta\bar{n}_{max}$.

It is possible to prove that for two ГТУ, carried out according to the same layout, with identical gas parameters but having different power and different design execution (different moments of inertia of the rotors and number of revolutions) this equality is true

$$\frac{\tau}{I n_0^2} = \Pi = \text{const}, \tag{32}$$

where Π is the parameter of pickup.

For two similar ГТУ of different power, dynamic stoking of revolutions will be identical

$$\Delta\bar{n}_{1max} = \Delta\bar{n}_{2max}. \tag{33}$$

Equations (32) and (33) permit (in necessary cases) producing a reduction of dynamic ГТУ characteristics.

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